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VIBRATION IN HYDROELECTRIC POWER PLANTS

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POWER DIVISION

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VIBRATION IN HYDROELECTRIC POWER PLANTS

Samuel Judd, M. ASCE[1]

SYNOPSIS

The vibration sources and design criteria employed by the United States Bureau of Reclamation in the structural design of its hydroelectric power plants are discussed.

INTRODUCTION

The many sources of vibration in a hydroelectric plant present a vexatious and controversial problem in every plant design. Some types of vibration that occur because of the forces of nature, such as wind or earthquake, play an important part in the proportions and economy of our structures. Others induced by moving or rotating equipment may cause loss of efficiency, objectionable noise, excessive wear and fatigue failures. Our interest in their presence, whether natural or man-made, is that they may result in a loss of property or loss of revenue—two subjects worthy of the concern of all of us.

Every object in motion produces vibrations that are exhibited in two forms —movement or noise. The form of vibration is dependent primarily on the frequency or the number of times the vibration repeats itself during a unit of time, usually a second. Dr. S. E. Slocum[2(1)] stated that when this frequency is below 30, it appears as a tremor that is apparent to the sense of touch. When it is higher, between 30 and 50, we perceive an intermittent noise, and when it lies between 50 and 10,000, it becomes a continuous sound.

Natural Sources of Vibration

The major sources of vibration resulting from the forces of nature arise from wind and earthquakes. Normally we must design our structures to resist these forces rather than control them as we may do with some of our destructive man-made vibrations.

The design of a hydroelectric plant for wind forces has been well established and is usually of secondary importance when the plant is in an area subject to earthquakes. All of the major quakes have resulted from the slippage of the earth's crust along a seam or fault. These tremors are termed tectonic quakes and they set up several types of elastic surface and depth waves. The location of these geologic faults or seams of weakness has been well mapped and may be obtained from the United States Geological Survey. These waves are very readily transmitted through saturated soils or rock since they propagate themselves in a manner similar to sound waves. In this

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2. Numbered references are listed at end of this paper.

respect it should be mentioned that the mere installation of a sand cushion beneath a structure is no cure-all, for the waves are readily transmitted through sand. The possibility of resonance between the shock from a surface wave and a power plant is remote because of the complexity of the earthquake motion; moreover, the inherent stiffness and shape of the plant result in a natural frequency considerably greater than that of the quake. Although the ground vibrations are not harmonic vibrations, certain paramount periods often appear, and this effect is usually included in the designs of the structure by the application of a force equivalent to a given fraction of the dead loads and fixed equipment loads. The fraction to be used for a given locality may be determined from the seismic probability maps prepared from historical records by the United States Geological Survey. For a stiff structure, such as a powerhouse, with a ratio of height to base width less than three, this method of design is satisfactory.

Man-Made Sources of Vibration

Our man-made vibrations are usually controlled through one or more of the following four methods, namely, isolation, elimination, damping, or absorbing the movement. To isolate the source of the vibration that arises in a hydroelectric unit is generally not considered feasible because of the weight and size of the component parts. The rotor of the generator for one unit at Grand Coulee, for example, weighs 500 tons and the turbine spiral cases are 50 feet in diameter. The elimination of the vibration, the most desirable method, by dynamic balancing or similar corrective measures, is never completely effective, because of physical and theoretical difficulties. The third method, damping, is one often used to reduce the effects of vibration, but its effectiveness is limited to a certain range of frequencies as will be illustrated later. The fourth method, absorbing or minimizing the displacements resulting from vibration, is most generally used because of the many sources and frequencies, some known and some unforeseen, that arise from the periodic man-made impulses.

Before discussing these causes and their treatment further, let us first review a basic equation portraying the effects of a forced vibration system with damping.[2] In such a system the equation for the actual periodic displacement is:

$$X_p = \frac{X_{st}}{\sqrt{(1 - r^2)^2 + (2\frac{C}{C_c} r)^2}}$$

Where X_p = periodic displacement

X_{st} = static displacement from a static load equal to the maximum value of periodically varying load

r = ratio of impressed frequency to natural frequency of the structure

C = actual damping factor of the system

C_c = critical damping factor of the system

It should be noted here that the actual periodic displacement of the structure can be made negligible by adding mass or stiffening the structure so that the static displacement is very small.

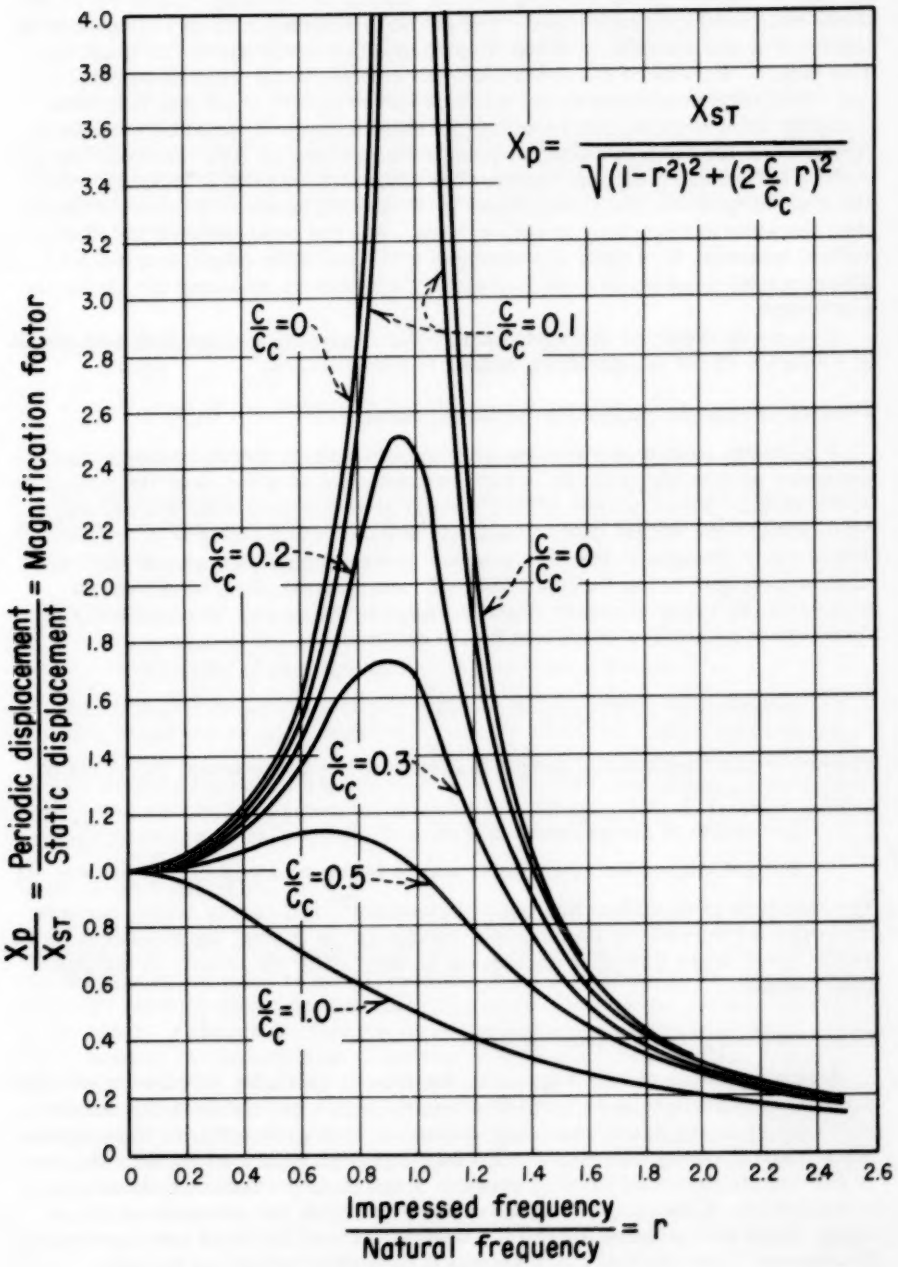

$X_p = \frac{X_{ST}}{\sqrt{(1-r^2)^2 + (2\frac{c}{c_c} r)^2}}$
$\frac{X_p}{X_{ST}} = \frac{\text{Periodic displacement}}{\text{Static displacement}} = \text{Magnification factor}$
$\frac{c}{c_c} = 0$
$\frac{c}{c_c} = 0.1$
$\frac{c}{c_c} = 0.2$
$\frac{c}{c_c} = 0$
$\frac{c}{c_c} = 0.3$
$\frac{c}{c_c} = 0.5$
$\frac{c}{c_c} = 1.0$
$\frac{\text{Impressed frequency}}{\text{Natural frequency}} = r$
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6 2.8 3.0 3.2 3.4 3.6 3.8 4.0

Fig. 1

The magnitude of this periodic vibration of the structure also depends upon the relation between the impressed frequency of the disturbing force and the natural frequency of the building and the amount of damping. Figure 1 shows these relationships graphically. The periodic displacement is represented in the ratio of the periodic to static displacement, usually called the magnification factor. As will be observed from the curves, if the impressed frequency and the natural frequency of the structure were nearly equal and very little damping were present, the resulting periodic movement would be very large. However, if the ratio of these frequencies is greater or less than one, the resulting movement varies as shown. The beneficial results of damping when the frequency of the disturbing force is 40 percent more or 40 percent less than the natural frequency is self-evident. For the remainder of the frequency ratios, however, it is seen that damping will have little effect, and the periodic displacement must be absorbed or made negligible by reducing the static displacement.

Now let us consider some of the periodic man-made forces that may result in vibration of the substructure and/or superstructure.

Periodic Pressure Surges Due to Water-hammer

A possible source of vibration in hydraulic turbine installations is the pressure surges which occur in turbine penstocks as a result of the operation of the turbine wicket gates. Under normal governor action during normal load changes the wicket gate movements produce rapid changes in the flow. These rapid changes in the flow produce water-hammer waves and rhythmic pressure surges in the turbine penstock. An example of these measured surges at the Tracy Pumping Plant is shown in Figure 2. The frequency of these pressure surges is defined by the relation:

$$F = \frac{V}{4L}$$

where F = the frequency of the pressure surges in the penstock in cycles per second

L = length of the penstock in feet, and

V = velocity of water-hammer waves in the penstock in feet per second

For this type of operation hydraulic resonance[3] will rarely occur, because the pressure surges are damped out rapidly by the partial wave reflections which occur at the partially open gates as well as by the effects of the pipeline friction.

Draft-tube Surges and Power Swings

A strong source of low frequency vibration in hydraulic turbine installations results from draft-tube surges. Draft-tube surges are the periodic surges which occur in the draft-tube water column of hydraulic turbines when operating at partial gate openings. The hydraulic phenomenon causing this surging is not completely understood at present[4] and does not occur at all turbine installations. However, it is believed to result from the presence of air or vapor which accumulates around the vertical axis of the draft tube just below the runner. The effect of these surges is described briefly as follows:

The fluctuation in the draft-tube vacuum or pressure causes a fluctuation in the effective head on the turbine and thereby causes a variation in the

turbine discharge. This produces a further fluctuation in the penstock pressure due to the effects of water-hammer. These in turn combine with the draft-tube surges to create a change in head on the turbine and a variation in torque on the runner. Since the turbine is connected to a generator, the electrical output of the machine also varies, and power swings occur. The magnitude of these power swings will then depend upon the relation between the frequency of the draft-tube surges and the natural frequency of oscillation of the generator connected to the power-transmission system. An example of these surges measured at Parker Power Plant is shown in Figure 3. The draft-tube surge frequencies and power swings have a rough relationship to the speed of the unit for various types and sizes of units. This relation is expressed empirically as:

$$F = \frac{N}{216}$$

where F = approximate frequency of draft-tube surges in cycles per second, and

N = speed of the turbine in revolutions per minute.

Guide Vane-runner Bucket Relation

Objectionable vibrations in a hydraulic turbine installation may be caused by having an improper relation among:

1) Speed of rotation
2) Number of guide vanes
3) Number of runner buckets
4) Length of the spiral case
5) Velocity of propagation of water-hammer waves.

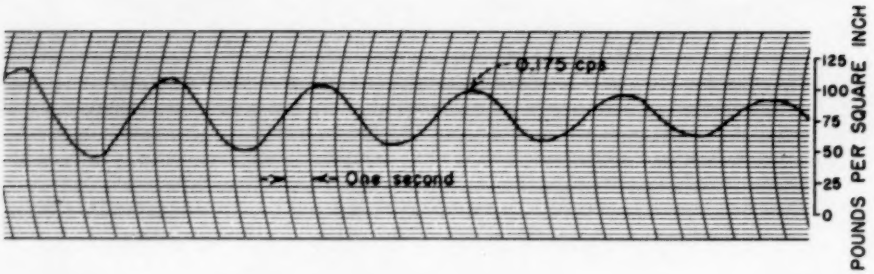
Den Hartog[5] describes a power plant in which serious vibration was apparently caused by an unfortunate combination of the variables listed above. The turbine was so proportioned that the pressure impulses caused by each runner bucket passing each guide vane were in phase, and as the various pressure waves reached the penstock, they added arithmetically to produce severe forced vibrations in the penstock.

Unbalanced Rotating Parts

Hydraulic turbine runners are usually balanced statically in the shop. This consists in mounting the turbine runner on a pivot and balancing the runner by the addition or removal of metal from the outer shroud band. On small generators the rotor is also balanced statically in the shop by the addition of weights in the rotor. Although the turbine runner and generator may each be in perfect static balance the combination is seldom in dynamic balance. This is due to the fact that during rotation, additional disturbing forces caused by centrifugal effects are imposed on the rotating unit. Dynamic balancing of the unit is usually obtained after rotating the unit at normal speed by attaching weights to the generator rotor. When dynamic unbalance is present the vibration is usually very pronounced. Its frequency is as follows:[6]

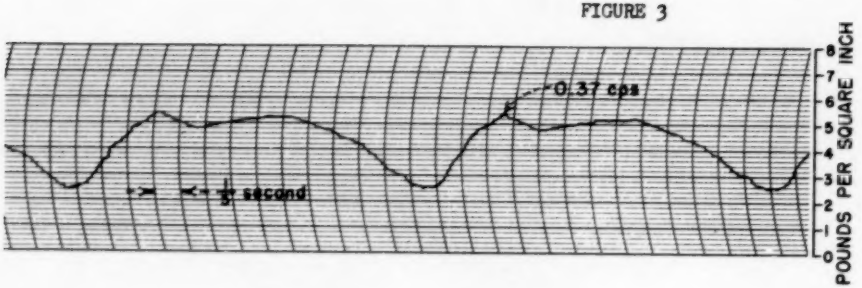
$$F = \frac{N}{60}$$

FIGURE 2



PRESSURE SURGES DUE TO WATER HAMMER
TRACY PUMPING PLANT
L = 5130 ft.
a = 3600 ft. per sec.

FIGURE 3



DRAFT TUBE SURGES
UNIT NO. 1
PARKER POWER PLANT
N = 94.7 R.P.M.
(0.50 gate position)

where F = frequency of vibration in cycles per second, and

N = speed of turbine in revolutions per minute.

Turbine Runner Vibration

Periodic vibration of turbine runner blades on large diameter runners (14 feet) has been observed at several turbine installations. This type of vibration has caused extensive cracking of the runner blades.[7] The cause of the runner vibration appears to be associated with the shock-type pressure changes at the trailing edges of the runner blades due to the presence of vortex trails. The action of these vortices imposes periodic transverse forces on the runner blades and causes them to vibrate across the stream lines. When this vibrating tendency is close to the natural frequency of the runner blades a condition of resonance occurs, and the vibration of the blades is magnified considerably. At Parker Power Plant, thinning of the trailing edge of the runner blades by chipping and grinding completely eliminated this type of vibration which could be heard and felt over the entire plant.

This type of vibration is also affected by the tail water elevation. At Keswick Power Plant the objectionable vibration disappeared when the tail water elevation was lowered 10 feet below the center line of the unit. Air admission at the lower seal chamber reduces the noise but has very little effect on the vibration within the unit. The range of frequencies as measured at Parker Power Plant was 55 to 65 cycles per second as shown in Figure 4, and that measured at Keswick Power Plant was from 150 to 170 cycles per second.

Pumps

Occasionally, a very strong vibration in pump installations results from the periodic pressure surges of high intensity which form inside the pump. While pump manufacturers do not seem to agree as to the exact cause of the phenomena, these pressure shocks appear to be the result of sudden velocity changes in the pump casing at the tongue of the pump. This seems to be borne out by the fact that insufficient clearance between the impeller and the tongue of the pump aggravates the pressure surges. These surges create shock waves which are transmitted through the water to the discharge line and pump casing. The measured pressure changes at the tongue of the pump at Grand Coulee Pumping Plant are shown in Figure 5. The frequency of this type of vibration is determined as follows:

$$F = \frac{Nn}{60}$$

where F = frequency of the pressure changes in the pump discharge in cycles per second

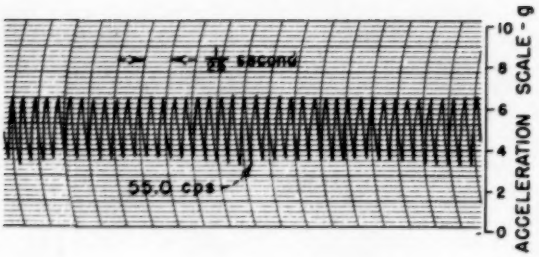
N = speed of pump in revolutions per minute

n = number of impeller vanes.

Design Criteria

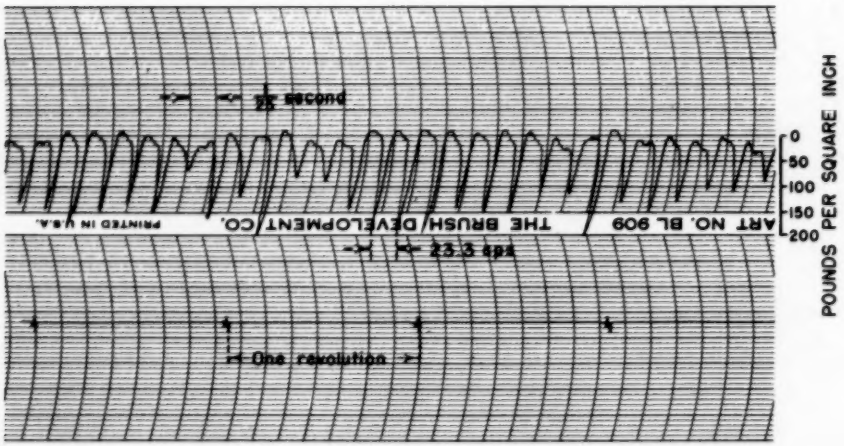
At present, we use several criteria in our design for controlling these man-made vibrations resulting from the aforementioned forces—whether it is vibration resulting from objectionable movement or noise. Briefly they are:

FIGURE 4



TRANSVERSE ACCELERATIONS OF TURBINE RUNNER
UNIT NO. 1 BLADE NO. 2
PARKER POWER PLANT
(0.8 gate position)

FIGURE 5



PRESSURE CHANGES AT TONGUE OF PUMP
GRAND COULEE PUMPING PLANT
N = 200
n = 7

a. An experience constant for the relation between mass and kilowatt capacity.

b. An experience constant for the ratio of the weights of the rotating parts to their concrete foundation.

c. Structural analysis for the impressed forces.

d. Mallock's nuisance vibration curve.

We have relied upon a minimum ratio of concrete mass to kilowatt capacity as a guide in proportioning our structures since the early thirties. While this is admittedly a very rough guide, it may be rationalized because certain losses are inherent in transforming the flow of water into electrical energy through mechanical means. Some of these losses are in the form of heat, some in friction, and some are vibration. As we have seen from the aforementioned examples the known sources of vibration are several and their frequencies may vary from 1.5 to 150 cycles per second (an unbalanced rotor and the turbine runner vibration, respectively, at the Keswick Power Plant). From Figure 1 it is obvious that we cannot rely on damping, even if its measurement were well defined, to control all vibrations because of its limited effective range. The energy lost in this transfer of kinetic to electrical energy is approximately 15 percent of the capacity of the hydroelectric unit. What part of this may be attributed to vibration is questionable, but undoubtedly small, probably less than one-quarter of one percent of the capacity of the machine. However, the energy lost due to vibration must be absorbed to minimize the displacement so that it will not be objectionable. To do this and cover the wide range of impressed forces and frequencies possible, we have used a minimum value of two for the ratio of the cubic feet of the concrete mass surrounding the scroll case and supporting the generator to the capacity of the machine in kilowatts. Many plants, both public and private, have a higher ratio than this. Using the data listed in Creager and Justin,[8] it will be seen that this ratio varies from less than 1 to more than 30. Other extenuating conditions, such as, high tail water or foundation conditions, undoubtedly have contributed to the wide variations.

We have searched for a reliable experience constant of this nature without success. Studies in which the mass has been varied with the rpm, head, discharge, specific speed, and horsepower have been attempted without reliable results. Unquestionably, one reason for such poor correlation is the lack of definite data on the proper mass to be used. The time and money required to assemble these basic data in the correct form have been prohibitive.

At the present we believe a more reasonable approach to an experience constant of this nature lies in limiting the acceleration due to a periodic force to a value between .005 g and .01 g.

A second criterion, the ratio of the weight of the rotating parts to their support is strictly based upon experience and, I believe, is similar to one used in steam plant design. The generator supports and scroll case encasement concrete are proportioned to minimize eccentricity and provide a minimum of two to two and one-half times the weight of the rotating parts.

The third criterion, structural analysis for the impressed forces, presents varying degrees of complexity. For periodic natural forces, wind or earthquake, the analysis is made along conventional lines with the impressed movement from a quake applied as a horizontal or vertical force whose magnitude is determined as equal to the mass times the acceleration. For other impressed forces arising from the sources previously mentioned the design

requires a determination of whether the whole structure or its components will be near resonance with any of these large periodic forces.

It is difficult to predict accurately the natural frequencies of vibration of the complex, massive, structures needed to house a hydroelectric unit. The method we employ in vibration analyses of power plants is to make simplified assumptions and idealizations such that the lowest possible natural frequencies are computed. If it can be shown that the computed frequencies are considerably higher than the highest of the impressed frequencies, the powerhouse will probably be free of major vibration trouble. If the computed natural frequency is less than one of the important impressed frequencies, a redesign of part of the structure may be necessary. If this redesign proves uneconomical, it is sometimes possible to make provision for changing the natural frequencies in the initial installation by such measures as a change from simple spans to continuous or to leave space for the future placement of concrete to assist in damping the vibration.

It may develop that part of the powerhouse structure is in resonance with the disturbance caused by a small unit such as a compressor, sump pump, blower, etc. This condition is usually not considered during the design of the powerhouse; however, if the resonant condition develops, it is taken care of by changing the natural frequency of the vibrating member or by mounting the disturbing unit in a vibration-isolating suspension.

Our fourth criterion and present guides to the limiting amount of acceleration permissible from one of the periodic forces is an English authority by the name of Mallock.[1] He found by experiment that a periodic vibration which produces an acceleration amounting to one percent of gravity is perceptible, and that when the acceleration reaches four to five percent of gravity, the vibration becomes distinctly unpleasant. His criterion for a nuisance vibration is stated as:

$$4\pi^2 x_o F^2 = .05g$$

where x_o = displacement

F = frequency

g = acceleration due to gravity

It should be remembered that power plant design is never predicated upon any of these criteria, that is, the lay-out is made with only the arrangement of the equipment and the dead and live loads and the foundation conditions in mind. Then the aforementioned criteria are used as a check or guide to a more preferable arrangement.

CONCLUSIONS AND RECOMMENDATIONS

In conclusion, we may say that it is believed that there is a definite ratio between the mass and properties of a unit which may be used as a guide for controlling these man-made vibrations, but it is still the green grass on the other side of the fence. Such a check or guide, of course, would be most beneficial during the preliminary planning stages of a plant lay-out.

We recommend to the Power Division, a systematic program of vibration measurement. Although much work has been done in this field on structures in general, to our knowledge there has been none specifically on power and pumping plants. Actual vibration records during normal operation of the units

as well as during starting and emergency shut-down periods should be obtained. Suitable locations would be on the concrete surrounding the draft tube, encasement of the scroll case, main support structure of the generator stator, and points on all floor levels as well as main columns in the superstructure. These measurements need not be done on all plants. On the contrary, they should be performed on a selected few—in particular the ones where vibration troubles are known to exist.

The vibration equipment which the Bureau now has available for such measurements are a vibration meter and vibration analyzer. The pick-up for the vibration meter is of the inertia-operated crystal type which responds to acceleration. Two stages of electrical integration are built into the vibration meter so that the velocity and displacement of the vibration can also be measured. The vibration meter by itself gives very little information about the frequencies involved. The vibration analyzer is therefore used with the vibration meter, and through tuned circuits, it determines the amplitude of the various components of the periodic vibration which are included in the total vibration. This equipment has a frequency range from 2.5 to 750 cycles per second.

In addition to the above measurements, which would give information on actual vibration frequencies and magnitudes, another series should be made to determine the natural frequencies of the structures in as many of the above locations as possible. This could be done by the use of variable speed vibrating machines using eccentric rotating weights.

Information of this type taken at similar points in selected plants, coupled with a program of harmonic and structural analyses, could remove the uncertainties connected with our present concepts and form the basis for an intelligent, analytical approach to the problem.

I wish to thank Mr. John Parmakian, Mr. William H. Wolf, and other Bureau of Reclamation engineers for their valuable assistance in preparing and reviewing this paper.

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c. Discussion of several papers, grouped by Divisions.
d. Presented at the Atlanta (Ga.) Convention of the Society in February, 1954.
e. Presented at the Atlantic City (N.J.) Convention in June, 1954.

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